ANALYSIS OF EXERGY PARAMETERS OF BIOGAS POWER PLANT

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Abstract. The techniques of an exergy analysis concerning various circuits of biogas units, which allows replacing traditional energy resources and improving environmental conditions, has been presented. The heat schemes of biogas units were proposed, and analysis of their effectiveness was made. The comparison of different cycle parameters of various biogas units (i.e. a combustion turbine unit, a combined cycle gas turbine unit with gas discharges into the boiler and a combined cycle gas turbine with a high-temperature vapor generator and a reheating stage) was made, and the comparison of their exergy characteristics was carried out. The results of exergy analysis had demonstrated that the cycle of biogas CCGT (combined cycle gas turbine) with a reheating stage and using a high-pressure steam generator is the most effective, that can be explained by the fact that the thermal energy proportions of combustion products, accounting for the steam cycle and the gas cycle are approximately equal, comparing to conventional combined cycle gas turbine units.
Keywords: exergy characteristics, biogas unit, combined cycle gas turbine, high-temperature steam generator, reheat of steam.

ANALIZA PARAMETRILOR EXERGETICI AI INSTALAȚIEI DE BIOGAZ

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Rezumat. Se prezintă metoda de analiză exergetică a diverselor scheme de realizare a instalaţii de producere a biogazului, care funcţional pot substitui instalaţiile energetice tradiţionale. Sunt propuse schemele termice tehnologice ale instalaţiilor de biogaz şi metoda de calcul al eficienţei acestora. S-a efectuat compararea parametrilor ciclurilor de funcţionare ale instalaţiilor examinate cu efectuarea calculelor şi analizei exergetice a caracteristicilor instalaţiilor cu turbine cu gaze, instalaţiilor cu turbine cu gaze şi abur (ciclul combinat) cu cazan utilizator şi a instalaţii cu turbine cu gaze şi abur cu generator de abur de înaltă temperatură și preîncălzire. Analiza caracteristicilor exergetice a demonstrat, că ciclu instalaţiei de biogaz cu preîncălzire intermediară a aburului şi cu generator de abur de înaltă temperatură este cel mai eficient ca urmare a distribuţiei aproape egale a energiei termice obţinute în ciclul de ardere între ciclurile de conversie realizate de către turbina de gaze şi turbina de abur, în comparaţie cu ce se observă în instalaţiile tradiţionale cu ciclul combinat.
Cuvinte-cheie: caracteristici exergetice, instalaţie de biogaz, ciclul abur-gaz, generator de abur de temperatura înaltă, preîncălzitor.

АНАЛИЗ ЭКСЕРГЕТИЧЕСКИХ ПАРАМЕТРОВ БИОГАЗОВОЙ ЭНЕРГОУСТАНОВКИ

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Аннотация. Представлена методика эксергетического анализа различных схем биогазовых установок, позволяющих заместить традиционные энергоустановки. Предложены тепловые схемы биогазовых установок и методика расчета их эффективности. Выполнено сопоставление параметров циклов рассматриваемых установок и приведен расчет и анализ эксергетических характеристик газотурбинной установки, парогазовой установки со сбросом газов в котел и парогазовой установки с высокотемпературным парогенератором и промежуточным перегревом пара. Анализ эксергетических характеристик показал, что цикл биогазовой энергоустановки с промежуточным перегревом пара и высоконапорным парогенератором является наиболее эффективным за счет примерно одинакового распределения тепловой энергии продуктов сгорания между газовым и паровым циклами, по сравнению с традиционной схемой парогазовой установки.
Ключевые слова: эксергетические характеристики, биогазовая установка, парогазовый цикл, высокотемпературный парогенератор, промежуточный пароперегреватель.

Introduction

Energy efficiency has been increasingly discussed around the world due to the perception of its contribution to many issues like energy consumption, carbon emissions and others. In this context, demand side management is the most common way for energy efficiency improvement and consequent energy saving [1]. In a changing scenario where renewable sources play an important role in energy supply, new approaches to energy efficiency may be very helpful on achieving a big amount of energy savings.

Although advantages of biogas power plants which are an alternative to traditional power plant, they have not received wide distribution yet [2].

First of all, this can be explained by the fact that in the case of using the natural gas, the costs of maintaining facilities are minimal, while the use of biogas involves difficulties with poorly predictable expenses for collection, transportation, storage and preparation of raw materials [3]. In such way substitution of traditional fuels by biogas is economically viable in areas that are located near objects of agricultural production where there are developed infrastructure of collecting and preparing biomass for further using at power plants [4].

Secondly, the expedience of using an alternative fuel is determined by thermal efficiency of power installation. One of the ways of increasing the thermal efficiency of biogas technologies is the use of combined-cycle power plants. For analyzing the effectiveness of various thermal schemes of biogas plants can be used the exergy method [5]. Exergy analysis is a universal method for evaluating the rational use of energy. It can be applied to any kind of energy conversion system or chemical process. An exergy analysis identifies the location, the magnitude and the causes of thermo-dynamic inefficiencies and enhances understanding of the energy conversion processes in complex systems [6]. In this paper, a various circuits of biogas power plants are analyzed using exergy analysis.

1. An exergy analysis of biogas units

Exergy of heat can be determined by calculating the maximum of specific work that can be obtained from the specific amount of disposable heat that's equal to the specific work of reversible Carnot cycle [7]:

$$l_c = q \cdot \eta_c , \tag{1}$$

where $\eta_c = 1 - \frac{T_0}{T}$ – thermal efficiency of the Carnot cycle;

q – specific amount of disposable heat, kJ/kg;

T_0 – temperature of heat receiver, K;

T – desired temperature of heat source, K.

Hence, the specific exergy of heat with the potential T:

$$ex = q\left(1 - \frac{T_0}{T}\right). \tag{2}$$

Exergy of the working fluid flow equals to the maximum of useful work that can be obtained in a reversible transition of the working fluid to a state of thermodynamic equilibrium with the environment (p_0, T_0):

$$ex = i - i_0 - T_0(s - s_0) = l_{flow} + \Delta ex , \tag{3}$$

where $l_{flow} = i - i_o$ – specific work of flow, kJ/kg;
$\Delta ex = T_0(s - s_0)$ – specific exergy losses in the working fluid flow, kJ/kg.
Because the work of working fluid flow is:

$$\Delta l_{flow} = -\,di = -\,vdp, \tag{4}$$

it's possible to define the specific work using *i-s* diagram:

$$l_{flow} = \Delta i_{1-a} = i_1 - i_a = -\int_1^2 vdp\,. \tag{5}$$

where: i_1 – i_a – the difference between the specific enthalpy of working fluid during expansion in turbine with production of specific work l_{flow}, kJ/kg.

For combustion products, that change their temperature while moving through the gas duct of steam generator, it is valid:

$$ex = T_o \cdot \Delta s, \tag{6}$$

where: $\Delta S = \int_{entrance}^{inlet} \frac{\delta q}{T}$ – reduction of the specific entropy of gas in during heat transfer to the working fluid (water), kJ/(kg·K).

Exergy efficiency of the cycle is determined by ratio of useful exergy Δexp_{us} to general consumed exergy Δex_{gen}:

$$\eta_c = \frac{\Delta ex_{us}}{\Delta ex_{gen}}\,. \tag{7}$$

For energy installations useful exergy is converted into actual work of cycle $l_{c.a}$ taking into consideration its irreversibility and general consumed exergy can be expressed by the difference of exergies:

$$\eta_c = \frac{l_{c.a}}{ex_1 - ex_2}\,. \tag{8}$$

By the difference of specific exergies or enthalpies at initial and terminal points of process, it can be determined the thermal efficiency of biogas power plant. Useful work at isentropic expansion of working fluid in a gas turbine can be defined as:

$$l_{gt} = e_{inlet} - e_{outlet} = i_1 - i_2, \tag{9}$$

where: e_{inlet}, e_{outlet} – specific enthalpies of working fluid at inlet and outlet, kJ/kg.

Lines of equal values of exergy at i-s diagram are straight. In the area of saturated vapor these lines coincides to lines T = const (P = const). The line at ex = 0 is tangent to isobar P_0 in environmental point 0. The line segment along isentrope, between the points, which define the state of matter, and the line of environment, represents the exergy concerning zero condition (fig.1).

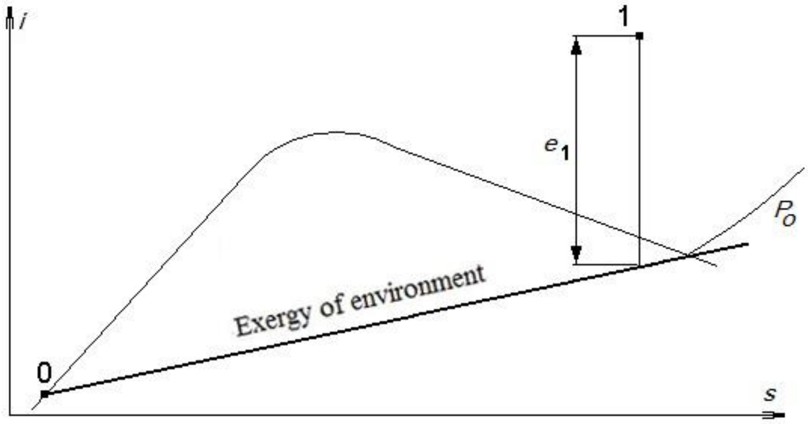


Fig. 1. *I-S diagram of water steam*
e_1 – exergy in point 1 concerning zero state

Specific work of isentropic expansion of gas in gas turbine:

$$l_{gt} = c_{p,g} \cdot \Delta t_{gt} . \tag{10}$$

where: $$\Delta t_{gt} = T_1 \left(1 - \frac{1}{\pi_{gt}^{\frac{k_g - 1}{k_g}}} \right), \tag{11}$$

$c_{p,g}$ – specific thermal capacity of working fluid, kJ/(kg·K);
T_1 – gas temperature at the inlet of turbine, K;
π_{gt} – degree of the gas expansion in the turbine;
k_g – adiabatic coefficient of the gas.
Specific work of isentropic air compression in the air compressor

$$l_k = c_{p,a} \cdot \Delta t_{ak} , \tag{12}$$

where: $$\Delta t_k = T_4 \left(\pi_k^{\frac{k_a - 1}{k_a}} - 1 \right) ; \tag{13}$$

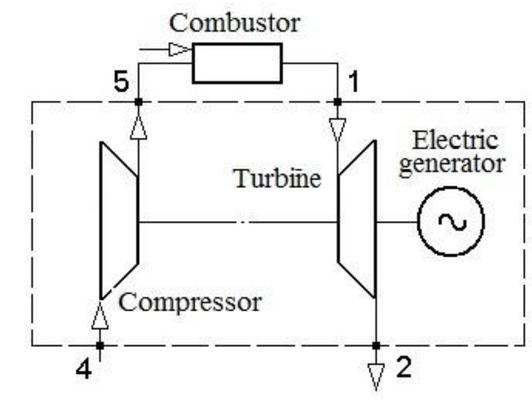
$c_{p,a}$ – specific thermal capacity of the working fluid (air), kJ/(kg·K);
T_4 – air temperature at the inlet of air compressor, K;
π_k – degree of the air compression in the air compressor;
k_a – adiabatic coefficient of the air.
Specific work of isentropic water compression in feed pump:

$$l_{pn} = \Delta p \cdot \Delta V , \tag{14}$$

where: Δp – pressure differential in the pump, kPa;

ΔV – specific volume of the feed water is supplied by feed pump to the steam generator m^3/kg.

2. The results of numerical simulation of the exergetic parameters
Exergy analysis method allows to determine the exergetic efficiency of various schemes of biogas power plants (Fig. 2 – 4) [8]. The results of calculations are in the Table 1.



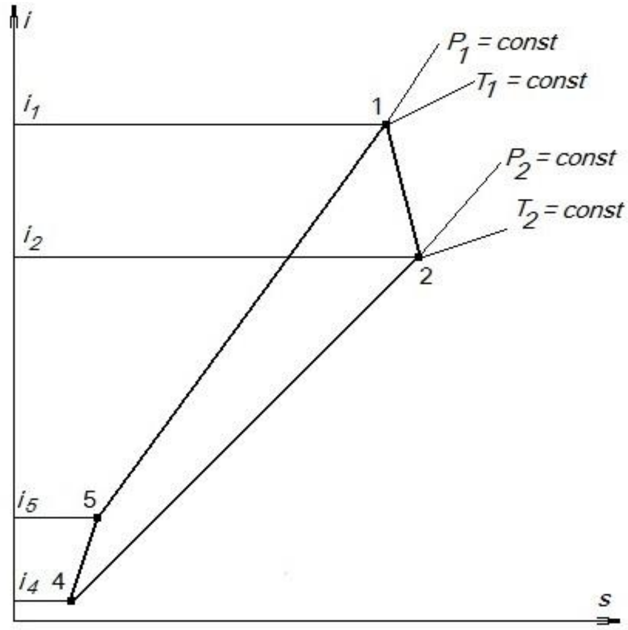


Fig. 2. *Scheme and the cycle of gas turbine unit*

Fig. 3. *Scheme and the cycle of combined-cycle unit with the gas discharge to the steam generator*

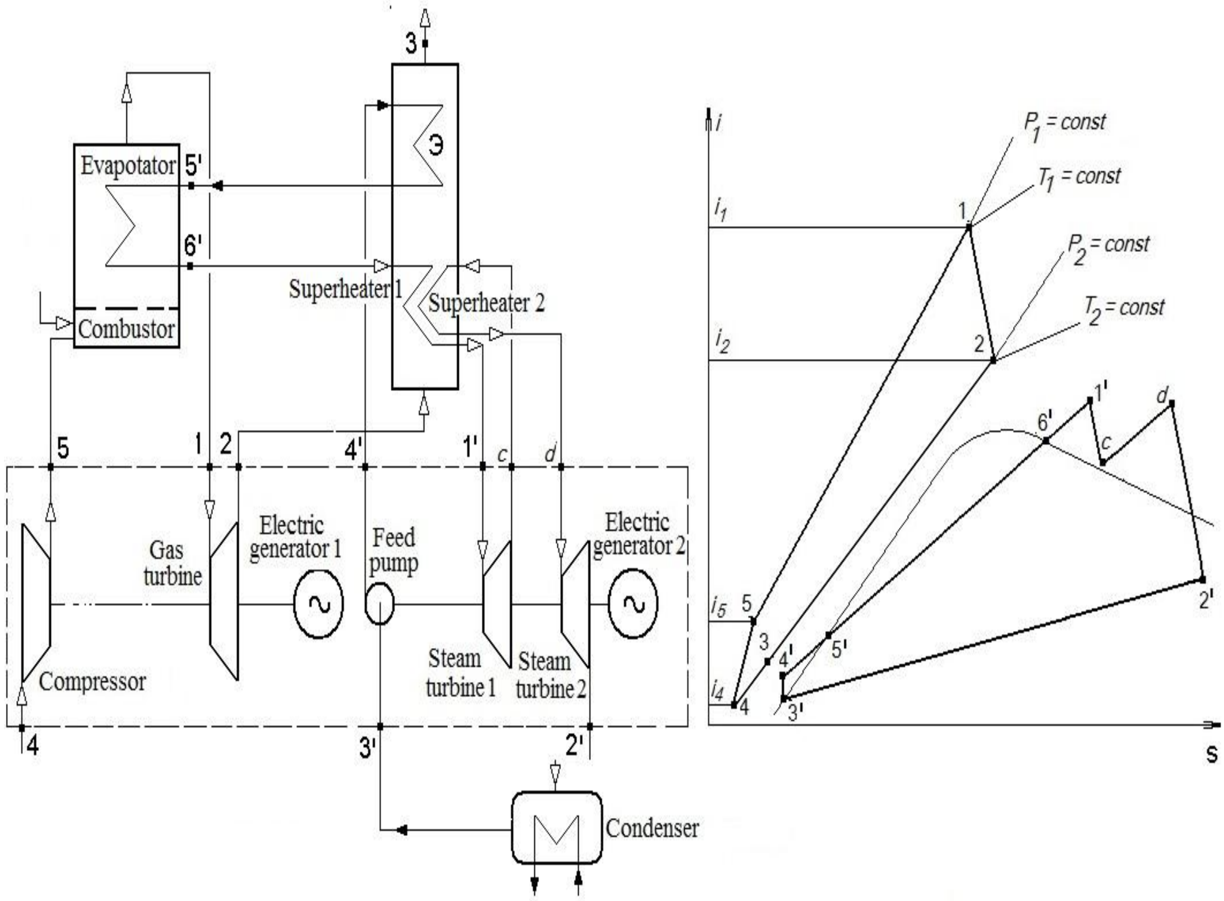


Fig. 4. *Scheme and the cycle of combined-cycle unit with the high-temperature steam generator and the intermediate superheater*

Table 1. The results of calculations for the different schemes of biogas plants

Fig.2	t, ^{0}C	P, bar	i, kJ/kg	Fig.3	t, ^{0}C	P, bar	i, kJ/kg	Fig.4	t, ^{0}C	P, bar	i, kJ/kg
1	1000	6,1	1250	$1^/$	550	10	3600	$1^/$	550	140	3480
2	588	1,02	807	$2^/$	33	0,05	2510	*c*	220	10,1	2880
3	20	1,0	21	$3^/$	33	0,05	138	*в*	550	10	3600
5	251	6,12	254	$4^/$	36	10,1	170	$2^/$	33	0,05	2510
								$3^/$	33	0,05	138
								$4^/$	36	141	170
								$5^/$	335	140,5	1531
								$6^/$	335	140,5	2592

Calculations show [9] that for the gas turbine $\eta_{c1=}$0,21; for combined-cycle unit with the gas discharge to the steam generator $\eta_{c2=}$0,37 and for combined-cycle unit with the high-temperature steam generator and intermediate superheater $\eta_{c3=}$0,47, that can be explained by the fact that proportions of the thermal energy of combustion products, accounting for the steam cycle and the gas cycle are approximately equal, comparing to conventional combined cycle gas turbine units.

Conclusions

Exergy analysis shows that that combined-cycle biogas plant with the high-temperature steam generator and intermediate superheater is the most effective (Fig. 4). It is explained by the fact that the proportions of heat combustion attributable to the steam cycle and to the gas cycle are approximately the same, in contrast to traditional combined-cycle plants.

In the traditional combined cycle plant it's possible to increase efficiency due to high potential gases at the outlet of the gas turbine flowing to the steam generator with further it's feeding to the steam turbine. However, proportions of the thermal energy of combustion products for steam cycle are about 3 times less than for gas cycle [10].

Heat scheme of biogas unit (Fig. 4) enables to correct this deficiency due to high-temperature steam generator, which realizes the same amount of thermal energy of the combustion products, as steam generator at the outlet of the gas turbine. Therefore, the proportion of the thermal energy of the combustion products consumed by steam cycle is doubled.

As a result of redistribution of thermal energy of the combustion products between the gas cycle and the steam cycle in favor of the steam cycle, as more efficient, the efficiency of combined cycle power plant increases in comparison with conventional scheme by 6%.

Results of numerical simulation shows that at optimal parameters of working fluid efficiency of biogas installation with the high-temperature steam generator and intermediate superheater increases by 10 % comparing to conventional combined cycle gas turbine units.

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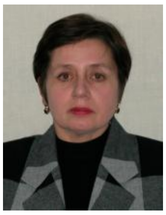

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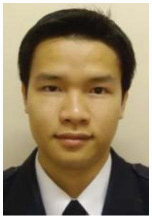

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